Series Fan-Powered Boxes:

Their Impact on Indoor Air Quality and Comfort

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Until the concerns about indoor air quality (IAQ) reached a peak in the last five years, interior zones of office buildings were traditionally served with cooling-only "shut-off" variable air volume (VAV) boxes. Heat gains from lighting and equipment, in addition to those from occupants themselves, caused supply air quantities of 13°C to 15°C (55°F to 60°F) air to be sustained at rates on the order of 2.5 L/s per square meter (0.5 cfm per square feet), a rate that most designers at the time felt would provide a comfortable and well ventilated environment. More recently, the reduction in lighting loads, the use of cold air systems, and, most importantly, the concern that acceptable indoor air quality may not be maintained at low air flow rates, has led to a different solution in interior zones: the series fan-powered mixing box.

The series box (shown in Figure 1 along with a parallel fan-powered and a cooling-only "shut-off" box) includes a supply fan in series with the VAV damper. The fan supplies a constant volume of air to the space consisting of primary air from the cooling system mixed with ceiling plenum return air. Series boxes can ensure that the space is supplied with a constant amount of air at any rate the designer chooses regardless of the supply air quantity required to meet thermal loads.

But use of fan-powered boxes has its disadvantages compared to shut-off boxes, including:

- increased first costs, both the cost premium for the boxes themselves plus the cost to wire them;
- increased maintenance costs due to the additional moving parts (the fans) and to change filters (if filters are installed);
- increased noise levels, both radiated from the boxes and conveyed through the duct system to air outlets;
- and substantially increased fan energy costs. If used on every zone in the system, series fan-powered box can reduce the fan power of the central air handler by reducing its total pressure requirement, but the power required by the

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Fan powered boxes much more than offsets this savings. This is true even for cold air supply systems (Ellison, 1993). Standard fan-powered mixing boxes have an overall efficiency (fan and motor) of about 15% compared to a combined efficiency on the order of 60% for central air handler fans. In addition, these fans run continuously at constant power, unlike central variable air volume fans which generally have variable speed drives or other unloading devices for efficient part load operation.

Do the benefits of fan-powered boxes in interior spaces offset these significant disadvantages? The purpose of this paper is to try to answer this question based on recent research and analysis.

**Perceived Benefits**

The arguments in favor of series fan-powered boxes in interior zones include:

1. They may improve ventilation effectiveness by improving air distribution performance, thereby improving IAQ.
2. They may improve air diffusion performance, providing more uniform temperatures and thereby improving occupant comfort.
3. They may increase air motion, which may improve IAQ and comfort.
4. If filters are provided in the fan-powered box, they can improve IAQ by removing particulate contaminants from the space.

5. For spaces that have a high ratio of required ventilation outdoor air to overall supply air, fan-powered boxes can increase the effective ventilation rate by transferring air from adjacent spaces that have a lower concentration of pollutants, thereby reducing the amount of outdoor air required directly from the supply air system.

Two other benefits (the ability of fan-powered boxes to improve heating mode operation by lowering supply temperature and increasing air flow rates through diffusers; and the ability of the fan-powered box to ensure ventilation is provided at low loads which can result in low primary air flow rates in shut-off VAV systems) are not listed above since these benefits may also be provided by parallel fan-powered boxes. Parallel fan-powered boxes do not suffer from the inefficiencies of series boxes because the box fans are sized for lower flow rates and do not run continuously. The issues analyzed here are the possible benefits unique to series fan-powered boxes applied to interior spaces.

Everyone has intuitive feelings about the validity of these perceived benefits, which is why many people are using series fan-powered boxes. The following is an objective analysis based on research rather than anecdotes.

**Ventilation Effectiveness**

Air change effectiveness, commonly referred to as ventilation effectiveness, is a measure of how well the ventilation system mixes the supply of outdoor air with room air. It may be determined in accordance with proposed
ASHRAE Standard 129P. A perfectly mixed space is defined to have an air change effectiveness of 1.0.

Contaminant removal effectiveness, also called pollutant removal efficiency, is a measure of the system’s ability to remove contaminants that are generated by sources within the space, such as from people or building materials. It is a function of the type of source (e.g. point versus uniform) and its location in the space and thus studies of pollutant removal efficiency are not as repeatable air change effectiveness studies.

When fan-powered boxes are used in lieu of cooking-only VAV boxes to handle the same cooling load \(Q_s\) at the same room temperature \(T_r\), the supply air temperature \(T_s\) increases as the supply air quantity \(V_s\) increases according to the relationship:

\[ Q_s \propto V_s(T_r - T_s) \]

For example, an interior zone may require 2 L/s m\(^2\) (0.4 cfm/ft\(^2\)) of 13°C (55°F) supply air from a cooling-only VAV box to maintain 23°C (73°F) in the space. If the series fan-powered VAV box is sized to maintain 5 L/s m\(^2\) (1 cfm/ft\(^2\)), then the supply air temperature, according to the equation above, must be about 19°C (66°F).

The higher air flow rate could increase air outlet performance, but the colder the supply air temperature, the greater the natural diffusion of supply air. The two factors have opposite effects, and according to research reports, they tend to counteract each other:

- Bauman et al. 1992 found that air that was supplied at 13°C (55.4°F) to 18°C (64.4°F) at between 2.5 L/s m\(^2\) (0.5 cfm/ft\(^2\)) and 4.7 L/s m\(^2\) (0.94 cfm/ft\(^2\)) had a similar air change effectiveness (all around 1.0). This suggests there is no value to increasing air flow if supply air temperature rises correspondingly. Short-circuiting of supply air only occurred when the supply air temperature was more than about 2.5°C (5°F) above room temperature, and then only slightly (effectiveness between 89% and 98%). The same configuration with colder air resulted in effectiveness equal to or slightly greater than 1.0.

- The same Bauman study also found that there was no significant difference in ventilation effectiveness (1.14 versus 1.12) when supply air flow was dropped from 3.0 L/s m\(^2\) (0.6 cfm/ft\(^2\)) to 1.0 L/s m\(^2\) (0.2 cfm/ft\(^2\)) using the same diffuser and the same 18°C (64.4°F) supply air temperature. This indicates that increasing air flow will not necessarily improve ventilation effectiveness even at the same supply temperature.

- In the cold supply air study by Zhang et al, 1994, it was found that the ventilation effectiveness and contaminant removal effectiveness were almost exactly the same, both near that for perfect mixing, for a high air flow rate at 13°C (55°F) and a low air flow rate at 3°C (38°F) using both square louver face and slot diffusers. It can be inferred that an even higher supply rate at 18°C (65°F) would have similar performance.

- Bauman et al 1993 concluded that “a ceiling mounted supply and return air distribution system supplying air over the range 0.2 to 1.0 cfm/ft\(^2\) (1.0 to 5.0 L/s m\(^2\)) was able to provide uniform ventilation rates into partitioned work stations. The range of tested supply volumes represented rates that were below and above the [diffuser] manufacturer’s minimum levels for acceptable performance.”

- Fisk et al 1991 found that “almost all measurements indicate very limited short-circuiting between locations of air supply and removal.” However, they found that the warmer the air, the worse the ventilation effectiveness became.

- Measurements by all major research to-date (e.g. Persily and Dols 1991, Persily 1992, Offerman and Int- Hour, 1989) indicate that air change effectiveness is around 1.0 for virtually all ceiling supply/return applications when supply air temperature is lower than room temperature.

- Fisk et al 1995 concluded that “when the supply air was cooled, the ACE [air change effectiveness] ranged from 0.99 to 1.15, adding to existing evidence that short-circuiting is rarely a problem when the building is being cooled.” This study was based on air flow rates ranging from 1.0 to 2.5 L/s m\(^2\) (0.2 to 0.5 cfm/ft\(^2\)) using linear slot diffusers as well as two types of inexpensive perforated diffusers. The pollutant removal efficiency was also found to be greater than or equal to 1.0 when in the cooling mode even at the lowest air flow rate tested. Consistent with other studies, as supply air temperature was increased above the room air temperature, the air change effectiveness decreased.

It is apparent that the warmer the air, the worse short-circuiting problems become. This data suggests that in the cooling mode, a fan-powered box will have no measurable impact on ventilation effectiveness.

**Thermal Comfort**

The comfort performance of an air distribution system is commonly measured by its air diffusion performance index (ADPI) in accordance with ASHRAE Standard 113-90. ADPI is the percentage of measurement locations in the occupied zone of a space where air temperature and velocity meet certain magnitude and uniformity criteria. An ADPI above 80% is considered acceptable.

Another means of evaluating system performance is to use thermal mannequins which simulate the heat transfer between humans and their environment. Heat loss data can be converted into a comfort index called Predicted Mean Vote (Fanger, 1970). A PMV in the range of -0.5 to +0.5 is generally considered to represent acceptable thermal comfort conditions.

The 1992 Bauman study found that air that was supplied at 13°C (55.4°F) at 2.5 L/s m\(^2\) (0.5 cfm/ft\(^2\)) resulted in an ADPI of 94%, while air supplied at 18°C (64.4°F) at 4.7 L/s m\(^2\) (0.94 cfm/ft\(^2\)) achieved an ADPI of 89%. The lower flow rate also provided a better comfort index, although both were within the acceptable comfort range. This suggests that raising supply volume with warmer air will make
very little difference and may even have a negative impact on comfort.

Zhang (1994) found similar results with cold air: the ADPI for both 13°C (55°F) and 3°C (38°F) supply air was around 97% for three different diffuser types. In fact the lowest recorded ADPI was for 13°C (55°F) air (the warmest tested) using slot diffusers, compared to 97% using roughly half the supply air flow for 3°C (38°F) air supplied through the same slot diffusers.

Bauman, 1995 found that ADPI fell when air flow rate in a VAV system supplying 13°C (55°F) air drops from 3.8 L/s/m² (0.75 cfm/ft²) to 1.3 L/s/m² (0.25 cfm/ft²), but the reduction in flow caused only a slight change in the PMV comfort index which remained within the comfort range. (This study did not include any tests to see what ADPI would have resulted if higher rates had been maintained with warmer supply temperatures.)

ADPI and air change effectiveness are closely related for ceiling supply systems (both are essentially measures of mixing efficiency) so the same forces come into play. Thus increasing air flow while simultaneously increasing supply air temperature can be expected to offer no measurable benefit to occupant comfort.

Air Motion

ASHRAE Standard 55-1992 states clearly that “there is no minimum air speed necessary for thermal comfort” if the other factors that affect comfort (drybulb temperature, humidity, mean radiant temperature, radiant and thermal asymmetry, clothing level, activity level, etc.) are within comfort ranges. We all experience this at our homes every day; how often are you sitting in a chair in your house, perfectly comfortable, with no air movement (windows closed, furnace off)?

Nevertheless, there is a common perception among many HVAC engineers that people desire air motion for comfort. It may be a psychological effect; they know the ventilation system is working if they can feel the air moving, much like the psychological benefit of the ribbon that can be found so often attached to the diffuser to provide a visual indication that the system is on. However, it is the author’s feeling that if the space is cool enough, this psychological need would disappear. This is supported by Fanger, 1974 who found that comfortable subjects had essentially the same skin temperature, suggesting that any combination of temperature and air motion that produces skin temperature in the proper range will produce comfort. Other studies (Berglund and Cain, 1989) have shown that the colder and dryer the room air, the “ fresher” people perceive the air regardless of actual outdoor air change rate.

Studies by Schiller, 1988 and Yamazaki, 1982 found that occupants expressed a desire for perceptible air motion, although neither study experimented with other operative variables (e.g. lowering room temperature) to see if the perception was truly related to air motion.

Most IAQ studies (summarized by Mendell, 1993) have found no correlation between low air velocity and IAQ complaints. However, some studies (Nelson et al, 1995) have found an association between the perception of low air motion and IAQ complaints, even though this perception may not relate to the physical measure of air motion. Again, this may be related to air temperature rather than air motion: people often perceive a space to be “ stuffy” when it is warm. It is difficult to imagine someone complaining of lack of air motion and stuffiness when a room is cool.

Whether air motion is a benefit or not is debatable, so the question is then: do fan-powered boxes increase air velocities in the occupied zone enough to be perceived? It must be realized that standard ceiling diffusers are intended to mix primary air with room air, and as the two air masses merge, air velocity drops. We consider the air diffuser good if it does this mixing without a perceptible draft in the occupied zone (e.g. without “dumping”). It is quite possible that the same effects that caused the fan-powered box to not affect ADPI and ventilation effectiveness will also cause it to not increase air velocity as well.

In fact, the 1992 Bauman study suggests that this is the case, although not all the data is presented in their report to directly support this conclusion. They found with various supply air flow rates and various supply air temperatures that velocity in the occupied space (in this case within partition workstations) differed by no more than 0.08 m/s (15 fpsm) which is almost imperceptible. In most cases, they concluded that velocity changes were so small as to be within the measurement error of the experiment. A later study by Bauman et al., 1995 found that at the same air flow rates, significantly higher air velocities were measured when supply air temperature was cold than when it was warm.

Int-Hout, 1993, concludes based on several studies that “ reducing air flows and increasing delta-t results in only a slight drop in measured air motion” and that “ it would appear that use of fan terminals (particularly series flow type) may not result in increased air motion over VAV "pinch-off" designs if proper diffusers are utilized.”

Filtration

If good filters were used on fan-powered boxes, there would be some benefit in space IAQ due to lower particulate concentrations. However:

- Fan powered boxes are generally specified without filters, particularly those without hot water coils, typical of those used in interior zones. (This contention is supported by the personal observations of the author as a contractor both in Atlanta and California, and by conversations with two of the largest suppliers of VAV boxes. No hard statistical data was obtained.)

- Where filters are installed at initial construction, because they are difficult to access above the ceiling, it is doubtful that they will be changed before they cause airflow rates to diminish. It is also quite likely, again because of their poor accessibility, that filters will not be replaced after the first change-out period.

- Even where filters are specified, they are generally 1" throw-away filters which are not efficient enough to be rated using ASHRAE Standard 52.1 dust-spot procedures (the minimum rating for which is 20%). These filters are designed to prevent large particles from clogging coils and will have virtually zero efficiency at removing respirable particles (those less than about 4 m). Most fan-powered boxes are not designed for filters with 30% dust-spot efficiency, the lowest required to have any appreciable efficiency for respirable particles. This is either because they do not have a filter channel wide enough to fit a low pressure drop 30% efficiency filter, or because the high pres-
Figure 2: Generalized Multiple Spaces Equation

The total outdoor air required for a system, \( V_{ou} \), is:

\[
V_{ot} = \frac{V_{ou}}{E_v}
\]

where

\( E_v \) = Overall efficiency of the ventilation system

\[ E_v = 1 + \frac{BX - CZ}{A} \]

\( A \) = Fraction of supply air to the critical zone from sources outside the zone

\[ A = F_p + (1 - F_p)F_r \]

\( B \) = Fraction of supply air to the critical zone from fully mixed primary air

\[ B = E_mF_p \]

\( C \) = Fraction of outdoor air to the critical zone from other, over-ventilated spaces

\[ C = 1 - (1 - E_{ac})(1 - F_p)(1 - F_r) \]

\( X \) = Uncorrected outdoor air quantity rate expressed as a fraction of primary supply air rate.

\[ X = \frac{V_{ot}}{V_p} \]

\( Z \) = Minimum outdoor air fraction to the “critical” space

\[ Z = \frac{V_{oc}}{V_{sc}E_{ac}} \]

\( E_m \) = Mixing effectiveness of the primary system mixing plenum relative to the critical zone, defined as the fraction of outdoor air in the primary supply air to the critical space divided by the average outdoor air fraction. The average \( E_m \) for all spaces is one, but it may be less or more than one for the critical space depending on how well the outdoor air and return air are mixed in the air handler and supply system before duct branches occur.

\( F_p \) = Fraction of the air supply to the critical space that is primary air. For single duct systems with no other recirculation paths other than the main system, \( F_p \) is one. For systems with multiple recirculation paths or multiple supply systems such as fan-powered terminal, dual duct, and dual conduit systems, \( F_p \) varies depending on the system design and, if \( V_{AC} \), the current operating condition.

\( E_{ac} \) = Air change effectiveness in the critical space, as determined by ASHRAE Standard 129P.

\( F_r \) = Fraction of directly recirculated air to the critical space that is representative of average system return air as opposed to return air from the critical space itself. For single duct systems, \( F_r \) is not relevant. For a system such as a dual fan/dual duct system, \( F_r \) is equal to one since the hot deck supply is fully mixed return air. For a system using local fan-powered boxes mounted in the return plenum, \( F_r \) will be a function of the physical location of the box in the plenum relative to the space served.

\( V_{sc} \) = Total supply air rate to the critical space (both that directly recirculated and that supplied from the primary system).

\( V_{oc} \) = Required ventilation outdoor air rate to the breathing level of the critical space.

\( V_{ou} \) = Uncorrected required outdoor air rate. This is equal to the sum of the required outdoor air ventilation rate for each space served by the system, adjusted for overall diversity in occupancy throughout the all spaces served by the system.

\( V_p \) = Total supply air of the primary system, adjusted for diversity for variable air volume systems.
sure drop of a 30% rated filter will cause the air flow to the
diffusers to vary significantly depending on how much air
is being drawn through the filters versus how much is being
supplied via the VAV system. So good filtration may not
even be an option with fan-powered boxes currently on the
market.

• Fan-powered boxes draw air from the ceiling plenum,
a space that is virtually never cleaned and in which dirt and
dust naturally accumulate. Particularly when filters are not
used on fan-powered boxes, their operation more than
likely will increase particulate concentrations in the occu-
pied space compared to conventional VAV systems
(although no data is available).

Hence, it is unlikely that fan-powered boxes will have a
positive impact on particulate levels and their associated
impact on IAQ.

Transfer Air

The use of transfer air for ventilation is based on the as-
sumption that dilution of contaminants in a space can be
achieved by ventilating with any air stream that has a con-
centration of contaminants that is lower than the desired
space concentration. The classic application of transfer air
is a toilet room. Odorous air is exhausted and replaced by air
transferred from adjacent spaces. Since the transfer air has
essentially a zero concentration of toilet room odors, it is as
effective as direct outdoor air for dilution of these odors.
The use of transfer air in such applications is recognized in
most ventilation codes and standards, including ASHRAE

Standard 62-1989 also specifically allows diluted return
air to be used in lieu of direct outdoor air supply through the
use of the so-called “multiple spaces equation,” Equation 6-
1. The equation, which applies to recirculating HVAC sys-
tems that serve more than one space, is based on the concept
that return air from spaces that are over-ventilated may be
used to ventilate other spaces. Unfortunately, the equation
in the 1989 Standard does not apply to fan-powered boxes
or other systems that have other than a single recirculation
path, such as a single duct supply system.

The concept has been generalized in the current draft of
the revised ASHRAE Standard 62, scheduled to be pub-
lished for public review this summer, so that credit may be
taken for systems with multiple recirculation paths such as
both fan-powered boxes and dual duct systems. With this
generalization, the “multiple spaces equation” (based on
Warden 1995) can be expressed as indicated in Figure 2.

As with the 1989 version, the equation requires that the
“critical space” served by the system be identified. This is
generally the space that requires the highest ratio of outdoor
air to overall supply air, such as an interior spaces. The most
extreme example is an interior conference room which is
densely occupied but otherwise has a low cooling load.

Although it is difficult to see by inspection due to its
complexity, this equation shows that increasing the amount
of recirculated or transfer air to the “critical zone” in a mul-
tiple space system will reduce the overall amount of outdoor
air that the system must supply. This suggests that using a
fan-powered box on the critical zone may then have a pos-
itve benefit on system energy usage by reducing the min-
imum outdoor air requirement.

But this only is true if the recirculated air is taken from
other non-critical spaces served by the system. This is quan-
tified in the variable $F_r$ in Figure 2. The further the box is
away from the critical space and the closer it is to the pri-
mary return air path, the higher the value $F_r$, approaching
one as the position of the box nears the return air entry to the
primary fan system. The closer the box is to the space
served, the more air is recirculated from the critical space,
and the lower the value of $F_r$. For fan-powered boxes lo-
cated directly over the space served near return grilles, the
value of $F_r$ will fall nearly to zero and offer little benefit to
ventilation.

Properly located, the effect of using a fan-powered box
to serve the critical zone is to lower the overall outdoor air
intake rate required. This is a good application for series
fan-powered boxes given that the alternative is to add a re-
heat coil and use a constant volume box to increase and
maintain primary air flow rates, an option that requires both
higher first costs and energy costs compared to a fan-pow-
ered box.

Since interior zones are often “critical” spaces for sys-
tems that serve both interior and exterior zones, outdoor air
intake might be reduced somewhat by using fan-powered boxes in all interior zones. However, this may not have a net
positive benefit in building energy usage, depending on the
local climate. The increased air flow rates will only allow
outdoor air to be reduced during warm weather. During cold
weather perimeter zones are generally the critical zones due
to the reduction in primary air flow to these spaces when op-
nerating in a heating mode, and to the higher percentage of
outdoor air that results when overall primary air flow rates
fall which improves interior zone ventilation. Thus the fan-
powered boxes will only reduce energy usage when outdoor
air temperatures are above return air temperatures, about
25°C (77°F) or so. But the inefficient fans in the series fan-
powered mixing boxes will incur an energy penalty during
every hour the system is operating. In most U.S. climates,
the number of hours when the weather is this warm are a
small percentage of overall operating hours.

Conclusions and Recommendations

There is a growing perception among HVAC designers
that using series fan-powered boxes to increase the air sup-
ply to interior spaces has a positive benefit on comfort and
indoor air quality. But this contention is not supported by re-
search studies. While the benefits of series fan-powered
boxes are questionable, there is little question that use of
these devices in interior zones increases first costs, main-
tenance costs, noise levels, and energy costs.

Currently available series fan-powered boxes are ex-
tremely energy inefficient, about 1/4 the efficiency of a cen-
tral fan system. Accordingly, the proposed revision to
ASHRAE Standard 90.1, published for public review this
past spring, severely limits their use. While this limitation
may not survive the public review process, it certainly sug-
gests that designers investigate other design alternatives be-
fore using series boxes.

The following design guidelines should be considered:

1. Only use series fan-powered boxes on “critical”
   interior zones, such as interior conference rooms. Locate
   the boxes well away from the return air grilles of the
   rooms served so that recirculated air mostly is composed
of return air from other spaces.

2. If interior zone loads are very low, rather than using series boxes to increase supply air flow rates, consider serving them with a separate air handler from that serving perimeter zones. This will allow supply air temperatures to be warmer without creating excessive air flow rates in perimeter zones. It will also reduce overall outdoor air intake (presuming the designer is complying with Standard 62 multiple space system requirements), reduce reheat energy losses in exterior spaces by allowing for a more aggressive reset strategy in cold weather, and it will enhance outdoor air economizer performance. Where separate air handlers are not possible or not practical, consider using a dual fan/dual duct system instead of fan-powered boxes. This system can be less expensive and more energy efficient (Warden, 1996).

References


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